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**SHOCK AND VIBRATION TESTS OF A
SNAP-8 TURBOALTERNATOR**

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ABSTRACT

A turboalternator for the SNAP-8 space power system was subjected to the expected vehicle launch vibration and shock loading in accordance with the SNAP-8 environmental specification. Subsequent disassembly and detailed inspection revealed some internal damage but the unit was judged to be operational.

SHOCK AND VIBRATION TESTS OF A SNAP-8 TURBOALTERNATOR

by

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SUMMARY

A turboalternator from the SNAP-8 space power system was subjected to the expected vehicle launch shock and vibration loads as described in the NASA SNAP-8 environmental specification. This turboalternator consists of a four-stage axial flow mercury vapor turbine coupled to a three-phase, 400 Hz, 80 kVA alternator. The two shafts are each mounted on polyphenyl-ether lubricated angular contact ball bearings.

After the shock and vibration testing at NASA-Lewis Research Center the turboalternator was sent to the Aerojet General Corporation for examination. Internal inspection revealed a decrease of the turbine rotor axial motion but not enough to prevent operation. This was due to galling-type damage causing restriction of movement of the outer bearing rings in the housing. The unit was still operational as bearing preload was maintained. There were several failures of external plumbing and instrumentation connections

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which are not flight-type hardware. The remainder of the unit was in excellent condition including the ball bearings which showed no brinelling. The alternator, although displaying some of the same outer ring to housing galling damage as the turbine, had lost no shaft axial motion or bearing preload and was fully operational and in otherwise excellent condition.

It was concluded after disassembly and inspection that the turboalternator was still operational after the environmental testing. It would, however, be desirable to plate, hard face, or Microseal the outer bearing ring outside diameters and mating surfaces in the housing to prevent the observed galling. There was no damage to the bearing running surfaces, either on the races or balls. The testing also pointed out that the flight-type external plumbing connections will have to be carefully designed and supported to preclude failure in the launch environment.

INTRODUCTION

As part of the SNAP-8 space power development program, the system components are required to demonstrate capability of enduring the simulated shock, vibration and acceleration loads that are expected during vehicle launch and maneuvering and as defined in the SNAP-8 environmental specification.

One of the components tested was the SNAP-8 turboalternator, unit 6/3. (Identified in previous literature as the turbine alternator assembly or TAA.) In each of the three major axes, the turboalternator was subjected to the specification values of sinusoidal and random vibration and the shock requirements. These shock and vibration tests were conducted at the Lewis

Research Center; the results of these tests were determined by a post-test disassembly, examination and evaluation of the turboalternator by Aerojet General Corporation (AGC), Azusa, California, as part of contract NAS5-417.

DESCRIPTION OF APPARATUS

Turbine-alternator assembly. - The turbine-alternator assembly consists of a four-stage cantilevered axial flow turbine coupled to a 400 Hz, three-phase, homopolar alternator. The turbine shaft is mounted on two 40 mm angular contact ball bearings and is coupled to the alternator shaft by a slender quill shaft running through the center of the alternator shaft. The alternator rotor is straddle-mounted on its own set of 40 mm angular contact ball bearings. Since the bearings are oil lubricated (by polyphenyl ether, 4P3E) and the turbine operates on mercury vapor, the two fluids are separated by the ''seal-to-space'', a system of visco and molecular shaft seals. A sectional view of this TAA is shown in figure 1. A more detailed description of the turboalternator is found in reference 1.

Because the turbine and alternator angular contact bearings are preloaded with a force approximately equal to the weight of the rotating masses [26-27 kg (55-60 lb)], there was particular concern about possible brinelling of the bearings especially if resonances in the axial direction occurred.

Facility. - The shock and vibration testing was conducted in the environmental test laboratory at the Lewis Research Center. Both the shock and vibration was done in the MB Electronics, model C210 vibration machine, which is capable of 124 500 newtons (28 000 lbf) at frequencies of up to 2000 Hz. This machine is shown in figure 2. The SNAP-8 turboalternator was mounted to the machine through a magnesium table base, and a rigid

aluminum mounting fixture. It was held by its trunnions in this fixture in the same manner as planned for the flight configuration. The turboalternator was restrained from tilting by a stiff over-the-housing bridge section with an attachment bolt to the alternator lifting lug hole. This setup is shown in figures 2 and 3. The magnesium table base was mounted on the vibration table. The combined weight of the turboalternator and its mounting fixture was approximately 410 kg (900 lb).

Instrumentation. - The accelerometers, located as indicated in figure 3, were installed as follows:

1. Alternator housing - three individual accelerometers were cemented to an aluminum block which was, in turn, cemented to the alternator housing on the vertical center line.

2. Alternator rotor - triaxial type mounted directly to the alternator rotor.

3. Control accelerometers - attached to the TAA mounting fixture. Three accelerometers were cemented to a 2.54 cm (1 in.) cube of aluminum. On the opposite side of the fixture, to verify the vibration inputs, three accelerometers cemented to a 2.54 cm (1 in.) aluminum cube were also used.

4. Turbine housing - three accelerometers cemented to an aluminum cube were located on the vertical center line of the turbine housing.

5. Turbine rotor - a triaxial accelerometer was cemented to a steel block which was, in turn, screwed to a tapped hole provided in the turbine rotor mounting bolt.

Visual readout of all accelerometers was provided and all vibration inputs and outputs were recorded on magnetic tape. An X-Y plotter for two selected accelerometers was also utilized.

DISCUSSION OF TESTING

Procedure

The previously described turboalternator mounting fixture was bolted to the vibration machine (shaker) table adaptor. The instrumentation was then connected, and a 1 G sweep was made to 2000 Hz on the fixture alone in order to determine fixture resonances. The turboalternator was then installed and the remaining instrumentation connected.

This particular SNAP-8 turboalternator was provided with an antirotation device which was inserted in the turbine shaft speed pickup access port. The purpose of this device was to prevent rotation of the shaft and consequent twisting and breakage of the accelerometer leads.

The sequence of testing was to complete all of the testing (sinusoidal, random, and shock) in one axis (Y axis first) and then rotate the fixture to test in the next (Z) axis. When this testing was complete, the entire shaker was rotated to place the mounting table above the shaker in order to test in the X axis. Figure 2 shows the shaker in this position. Figure 4 shows the location and direction of the three test axes.

No operational tests of the turboalternator were performed during or after this testing because of the complexity of a test rig that would be capable of yielding meaningful results. However, after completion of these tests and prior to shipment to AGC, the rotational torque of the TAA shaft was measured with the space seal elements in the normal operating position. This isolates bearing torque and will determine if gross bearing damage was present, and also establish the bearing torque prior to shipment to AGC. A recording accelerometer was included in the shipping container to monitor the shock

loads incurred during shipment, both prior and after the shock and vibration testing at Lewis Research Center. In this way, it could be positively ascertained that the shipping loads did not exceed those during the test series.

Testing in the Y Axis

Sinusoidal vibration. - The first series of tests were conducted shaking in the direction of the Y axis. A summary of the inputs and responses for the entire test series is contained in table I. A typical response accelerometer printout is shown in figure 5. During the sweep with the fixture alone (run 1), significant resonances indicated by the fixture accelerometers, were noted with peaks to $7\frac{1}{2}$ G at 270 Hz in the X axis and 13 G at 1600 Hz in the Z axis with many other significant peaks distributed in this frequency range. The total time for this and all sinusoidal scans was 4 minutes.

With the TAA installed, although not a part of the SNAP-8 environmental specification, a sinusoidal sweep to 2000 Hz at 1 G input was made (run 2). During this test, the control accelerometer showed an increase in G input to about 1.8 G at 160 Hz with about 1.4 G over a band from 120 to 200 Hz. This accelerometer also showed a number of peaks to as high as $2\frac{1}{2}$ G from 500 to 1900 Hz. During this scan, some significant peaks were noted on the X and Z axis turbine rotor accelerometers. The largest of these occurred at 190 Hz and recorded 17 G. During this test, the Y axis turbine rotor accelerometer was reading erroneously.

The alternator rotor accelerometer showed a large peak in the Z axis of 15 G at 190 Hz and 38 G at 1150 Hz. In the Y axis, peaks of 3.3 G at 190 Hz and 3.1 G at 1200 Hz were recorded. In the X axis, the major peaks were 10 G at 172 Hz and at 670 Hz and 32 G at 1200 Hz.

Random vibration. - A random vibration was imposed in accordance with the specification. The random excitation was to be conducted for 3 minutes to the power levels as follows:

20 - 100 Hz at 3 decibels per octave (dB/Oct) increase

100 - 600 Hz at $0.4 \text{ G}^2/\text{Hz}$

600 - 2000 Hz at 6 dB/Oct decrease

The total G value overall for this specified power level is calculated to be 19.7 G rms. However, the power limitations restricted the overall power level during testing to about 13 G rms. During the test (run 4), peak values on the Y axis accelerometers of 40 to 50 G were noted at the completion of the test. It was observed that one of the turbine bearing lubrication lines which connects the housing fitting to the lubricant manifold was broken and the manifold was therefore removed at the end of this test.

Shock. - The next test consisted of three shocks of 15 G in the Y axis in each direction. Runs 5, 6, and 7 were in the +Y direction and runs 8, 9, and 10 were in the -Y direction. Response data were not recorded.

Testing in the Z Axis

Sinusoidal vibration. - A sinusoidal 1 G scan to 2000 Hz (run 11) was conducted over a 4-minute period of time. Peaks of 90 G at 220 Hz and 30 G at 490 Hz were noted for the Z axis accelerometer on the turbine shaft. Peaks of 50 G at 220 Hz and 20 G at 500 Hz were noted on the alternator Z axis accelerometer (fig. 5). In view of these large magnifications, it was decided not to run a sinusoidal scan test at 2 Gs in the Z axis. The SNAP-8 environmental specification requires only 0.25 Gs peak from 4 to 35 Hz for 3 minutes.

Random vibration. - The random vibration test was conducted in the Z axis at the same power level as described previously for the Y axis (run 12). Again, the exciter power limited the overall input to about 13 G rms. The largest G levels noted during the random tests were 70 to 80 G for the Z axis accelerometer on the alternator rotor and 80 to 100 G on the turbine rotor Z axis accelerometer. It was later determined however, that this accelerometer on the turbine was reading erroneously during this test. At the conclusion of the Z axis random test, it was noted that the turbine space seal manifold was broken at the weld joints of the three upper tube-to-housing joints. The lower tube connection, however, remained precariously intact. Before continuing with the shock testing in this axis, the space seal manifold was removed from the TAA. It was also noted that a 6-inch-long cantilever section of the thrust balance tube which was attached to the turbine exhaust housing was also cracked approximately two-thirds of the way around its weld joint. This was also physically removed before proceeding with the shock test.

Shock. - For runs 13 to 18, the TAA was next subjected to three shocks of 15 G in each direction in accordance with the specification. Data were not recorded for these runs.

Testing in the X Axis

Sinusoidal vibration. - The X axis sinusoidal scan (run 19) was made at 1 G uneventfully. In general, responses were of a lower magnitude than in the Y and Z axis sinusoidal tests.

Random vibration. - The random test (run 20) was then conducted in a similar manner to that described previously. The maximum peaks on the rotor

accelerometer for the turbine and alternator showed from 30 to 40 G in this test. At the end of the random tests, it was observed that one of the turbine pressure taps next to the exhaust manifold was cracked at the weld joint.

Shock testing. - The TAA was then shock tested at 15 G for three shocks in each direction in the X axis. These runs (21 to 26) concluded the TAA vibration and shock testing.

Following these tests and prior to shipment to AGC, the rotational torque of the TAA shaft was tested with the space seal elements in the normal lifted position. The torque was within specification and approximately the same as prior to vibration testing indicating no gross bearing damage.

RESULTS - POST-TEST EXAMINATION

External Examination

Upon arrival at Aerojet General Corporation, the shipping case accelerometer showed no significant shocks had been experienced during transit.

An external examination of the turboalternator shown in the as-received condition in figure 6 revealed no failures of the basic structure. Some of the external instrumentation and plumbing connections failed during the tests, but these were confined to nonflight parts such as the space seal manifold, thrust balance tube, and a lubrication plumbing manifold tube. The seal cavity pressure tap was also cracked at the root.

After removal of the antirotation device, the turbine and alternator shaft assembly rotated at the pretest torque value with the seals lifted off the shaft runner.

Internal Examination

Turbine. - The turbine was separated from the alternator. The drive spline teeth, on alternator and turbine were in excellent condition. The nuts on the turbine mounting pad joint untorqued evenly in the normal range of assembly values.

The turbine rotor group rotated with 0.795 newton-meters (7 in. -lb) of torque. The total rotor axial motion was found to be 0.0889 millimeters (0.0035 in.) with the neutral position of the rotor toward the turbine rather than the normal position toward the alternator.

The inlet housing nuts were removed at normal torque values and the labyrinth seal slid freely over the 1st stage wheel hub. A light intermittent pattern of labyrinth contact marks was noted on the first stage wheel hub (fig. 7). This unit, 6/3, is equipped with fixed style labyrinth seals. No other contacts were noted. No chatter or vibration marks were noted on the K-seals or the inlet to turbine case flange joints.

The locking device for the turbine bolt was secure and when it was released, the removal torque for the turbine wheel bolt was 6.83 newton-meters (600 in. -lb), which was identical with the assembly value. The bolt head and threads showed no vibration marks. The 1st stage turbine wheel showed a very slight pattern from contact with the labyrinth on the downstream side hub, similar to that on the upstream side. The second stage nozzle was normal in every respect. The second and third stage wheels also had a light labyrinth contact pattern on the upstream hub. No other marks were found. A small amount of fine metallic debris was found on the bottom of the turbine case under the third- and fourth-stage nozzles.

No definite point of origin was established at this time in spite of considerable effort. A sample was taken and upon analysis it was shown to be nine chrome 1 molybdenum (the housing material) with small but measurable amounts of aluminum and silicon.

The third-stage nozzle assembly was completely free of marks or damage. This stage wheel, including the hubs, was free of marks. The fourth-stage nozzle assembly and wheel were completely unmarked or undamaged. All stages showed good contact patterns in the curvic couplings with no evidence of damage or loosening of the clamping.

The visco pump clearance was 0.127 to 0.152 millimeter (0.005 to 0.006 in.), identical with the assembly value. The visco pump nut untorqued at 90 percent of buildup valve. No vibration damage was noticed on either the nut or visco pump ring.

The attachment screws for the turbine housing support arm mount had loosened. Assembly values were 85.2 newton-meters (75 in. -lb) and dis-assembly values 51.2 newton-meters (45 in. -lb). In addition six out of eight screws holding the seal housing to the bearing housing had lost 50 percent of their torque. The taper pins which actually aline the turbine case were still tight.

Rotor rotational torque was 5.54 newton-meters (78 oz-in.) with only one seal in contact after the turbine case was removed. The running ring was removed thus disengaging the face seals. A torque of 0.1775 newton-meters (2.5 oz-in.) was then measured which reflects bearing friction only. The removal of the rotor group from the bearing housing was difficult. The normal sliding fits between bearing outer rings, bearing lube rings, the spacer

sleeve and the bearing housing had become damaged and galled. This is shown in figures 8 and 9.

The rotor group slinger nuts were tight and no further evidence of damage were noted. The preload springs were undamaged and within normal tolerances for load calibration.

The shaft and slingers were in good condition. The bearing housing bore was in poor condition with many scratches and galls particularly at the alternator drive end (fig. 9). All significant parts of the turbine were fluorescent penetrant inspected to supplement the visual inspections of the parts and ensure discovery of damaged areas. A crack was found at the root of a pressure tap line on the inlet housing by this method.

A helium leak check of the bellows liftoff actuator for the start seals showed no leakage.

The turbine end bearing was in excellent condition except for several small patches of fretting corrosion and a large gall area on the outer ring outside diameter. The inner raceway showed a superficial chatter pattern at ball spaced intervals with the normal contact angle. No brinelling damage was present on either the raceways or balls.

Figure 10 shows the disassembled turbine bearing for the alternator end. The alternator end bearing was in good condition except for three gall areas on the outer ring outside diameter originating at the thrust face edge; this is not readily visible in figure 10. The outer ring raceway only had a light ball vibration pattern marking at 0 radians (0^0) contact angle with normal ball spacing between the patterns.

Alternator. - The mechanical inspection of the assembled alternator indicated the alternator was in good condition. The shaft axial motion was identical with the 0.3175 millimeter (0.0125 in.) measured at buildup. Rotor rotational torque was identical with the 2.845 newton-centimeters (4 oz-in.) measured at buildup. The spline damage was negligible. Comparison of dimensions over pins before and after vibration showed no wear. However, a helium leak check showed a leak several orders of magnitude greater than the specification limited of 1×10^{-7} standard cubic centimeters per second, at the metal to ceramic joint of the neutral terminal. In addition the alternator heat exchanger preservative fluid was noted to contain suspended particulate matter. Semiquantitative emission spectrographic analysis of the residues showed this to be rust.

The disassembly of the alternator proceeded normally until the end bell at the antidrive end was removed. The bearing and the lube ring came off with the end bell, even though the bearing inner ring to shaft fit is tight. When the bearing was eventually pressed out of the end bell, the outer ring outside diameter and the bearing stop were found to have galled and scratched the housing seat. This is shown in figures 11 and 12. Each damaged area originated with material transfer from the end bell bearing seat to the outer ring of the bearing. There was no noticeable loss of material from the bearings. When the drive end bell was removed, the drive end bearing and bearing stop were marked in similar fashion. No damage was found to the preload springs and they retained the same load characteristics as at assembly.

The outer ring of the alternator drive end bearing had a large amount of metallic pickup in one area. In addition there were smaller spots of pickup and small areas of fretting corrosion (fig. 13, not readily visible). The thrust

face had light chatter marks scattered over the surface. The raceway was undamaged, but showed a few light ball spaced stains. The separator had a very light contact pattern on the outside diameter and in the ball pockets. The inner ring bore had light axial scratches. The ring faces were clean and the raceway showed ball spaced contact at the normal contact angle. The marking was superficial. No brinelling damage was present on either raceway or balls. The balls were all in excellent condition.

The outer ring outside diameter of the alternator antidrive end bearing had one large, several medium, and many small sized galls and pits. The ring faces were clean. The raceway had several ball spaced chatter marks at zero degrees contact angle, as shown in figure 19. The separator had very light contact marks on the outside diameter and light contact marks in the ball pockets. The pocket contact patterns were heavier than those in the other bearings, both turbine and alternator.

The inner ring bore and faces were unmarked. The raceway showed two ball spaced contact patterns in the normal position. The balls were all in excellent condition, with no brinelling damage.

General Evaluation

The bearing outer ring to housing damage explains the anomalies noted in the teardown inspection. These are reduction of rotor axial motion from 0.127 to 0.0089 millimeters (0.005 to 0.0035 in.) and the shifting of rotor position towards the turbine in lieu of the normal position towards the alternator under the action of the stiffer spring. With the lube ring or the bearing outer ring at the alternator drive end frozen in the housing, only the action of the spring at the opposite end would affect the rotor thus pushing the rotor

towards the turbine. The galling or sticking of the elements in the housing would not permit the sliding of the outer ring at the alternator end which is necessary in order to allow the entire shaft motion. When the rotor group, including the nonrotating elements, were removed from the housing the rotor had shifted to the normal position towards the alternator under the effect of the stiffer spring.

The marking of the turbine wheel hubs from contact with the labyrinths clearly shows the limit of deflections of the shaft. The marking is superficial in nature and is found on both the first and second stage wheels with a uniform distribution around the periphery. This minor contact is apparently sufficient to dampen the vibration without serious impact effects. It should be noted that the lightly clamped labyrinth designs, such as used on previous turboalternators which are employed normally, would not necessarily act in a similar fashion but would be deflected under the vibration and shock impact forces.

The minor debris found in the turbine case was not traced to a specific area. However, the fabrication of the turbine case would afford many possibilities for particles to accumulate in the pressure taps, three of which are blind tubes. The cleaning procedure for this part does not include ultrasonic cleaning due to the size of the part which precludes accommodating within the ultrasonic tank. Therefore, it is possible that the loose matter was in the pressure tap tubes at the time of assembly.

CONCLUSIONS AND RECOMMENDATIONS

The following conclusions and recommendations were drawn from the environmental tests of the SNAP-8 turboalternator:

1. Reduction in turbine shaft axial motions was caused by sticking of the outer ring in the housing due to vibration-induced motion and consequent galling. The turbine and alternator were still functional since the bearing spring preload was maintained. To alleviate this condition, it is recommended to rework the bearing journals in existing turbine and alternator hardware by plating, hard facing, or by Microsealing to prevent fretting and galling. The same applies to the outer ring of the bearing.

2. Redesign of lubrication and coolant plumbing for flight configuration should emphasize integration within the housings such as drilled passages. External plumbing must then be rigidly supported.

3. The ball bearings are considered to have displayed acceptable performance in view of the lack of brinelling damage. Improved bearing performance should result when the outer ring to housing sticking problem is resolved and the preload will therefore be more consistent.

4. The alternator terminal ceramic to metal seal which showed increased leakage, is not critical in a flight machine. The exterior vacuum in service would result in practically no differential pressure across the deficient joint. Therefore the leak is classified as a minor problem.

5. The testing also identified the critical resonant frequencies of the turboalternator. Consideration to dampening these frequencies should be given for a flight-type mounting design.

REFERENCES

1. Cherry, M. G.; and Oiye, G.: Design and Development of a Turbine-Alternator Assembly for the SNAP-8 System. AIAA Specialists Conference on Rankine Space Power Systems. Vol. 1. AEC Rep. CONF-651026, vol. 1, 1965, pp. 477-497.

TABLE I

| Run description | Run number | Test axis | input amplitude | input frequency | Sweep speed | Run duration | Maximum responses | | | |
|------------------------------|------------|-----------|---|-------------------------------|-------------|--------------|----------------------|----------------------------|------------------|--|
| | | | or G level | Hz | oct/min | min | G level | Frequency Hz | Axis | Location |
| | | | in. doub. ampl. | Hz | oct/min | min | | | | |
| Fixture survey | 1 | Y | 0.25 1 G | 5-9 9-2000 | 4 | 2.15 | 13 | 1600 | Z | Trunion mount |
| Sinusoidal vibration | 2 | Y | 0.25 1 G | 5-9 9-2000 | 4 | 2.15 | 17 38 | 190 1150 | X&Z Z | Turbine shaft Alternator shaft |
| Sinusoidal vibration | 3 | Y | 0.25 2 G | 5-13 13-2000 | 4 | 2.15 | 28 40 40 | 200- 230 1150 | Y Z Z | Turbine shaft : Alternator shaft |
| Random 13g rms overall | 4 | Y | +3 dB/octave .4 g ² /Hz -6 dB/octave | 20-100 100-600 600-2000 | | 3 min | 40-50 | ---- | Y | Turbine and alternator shaft |
| Shock | 5, 6, 7 | +Y | 15 G | | | 11 millisec | | | | |
| Shock | 8, 9, 10 | -Y | 15 G | | | 11 millisec | | | | |
| Sinusoidal | 11 | Z | 0.25 1 G | 5-9 9-2000 | 4 | 2.15 | 90 30 | 220 490 | Z Z | Turbine shaft |
| Random 13g rms overall | 12 | Z | +3 dB/octave .4 g ² /Hz -6 dB/octave | 20-100 100-600 600-2000 | | 3 min | 50 20 80 | 220 500 ---- | Z Z Z | Alternator shaft : Alternator shaft |
| Shock | 13, 14, 15 | +Z | 15 G | | | 11 millisec | | | | |
| Shock | 16, 17, 18 | -Z | 15 G | | | 11 millisec | | | | |
| Sinusoidal | 19 | X | 0.25 2 G | 5-13 13-100 | 6 | 1.5 | 28 28 25 28 | 375 1000 340 1450 | X Z X Y | Turbine shaft Turbine shaft Alternator shaft Alternator shaft |
| Random 13g rms overall | 20 | X | +3 dB/octave .4 g ² /Hz -6 dB/octave | 20-100 100-600 600-2000 | | 3 min | 30- 40 | ---- ---- | X | Turbine and alternator shaft |
| Shock | 21, 22, 23 | +X | 15 G | | | 11 millisec | | | | |
| Shock | 24, 25, 26 | -X | 15 G | | | 11 millisec | | | | |

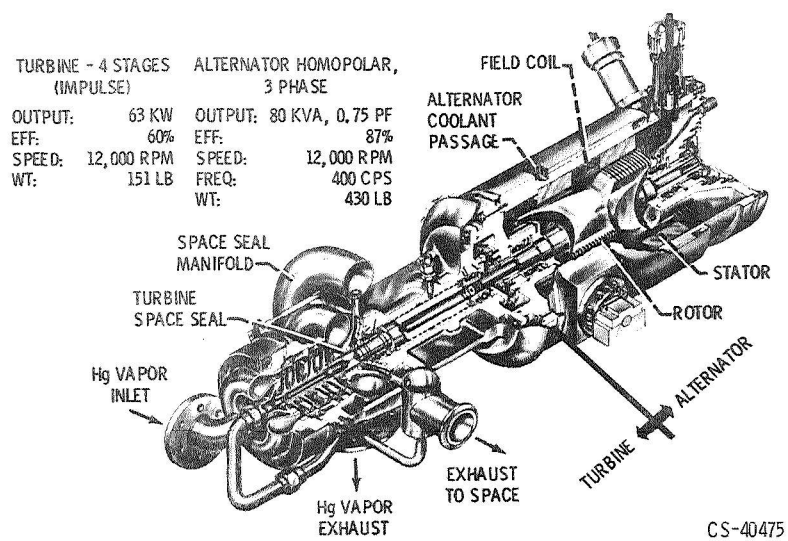


Figure 1. - SNAP-8 turbine alternator assembly.

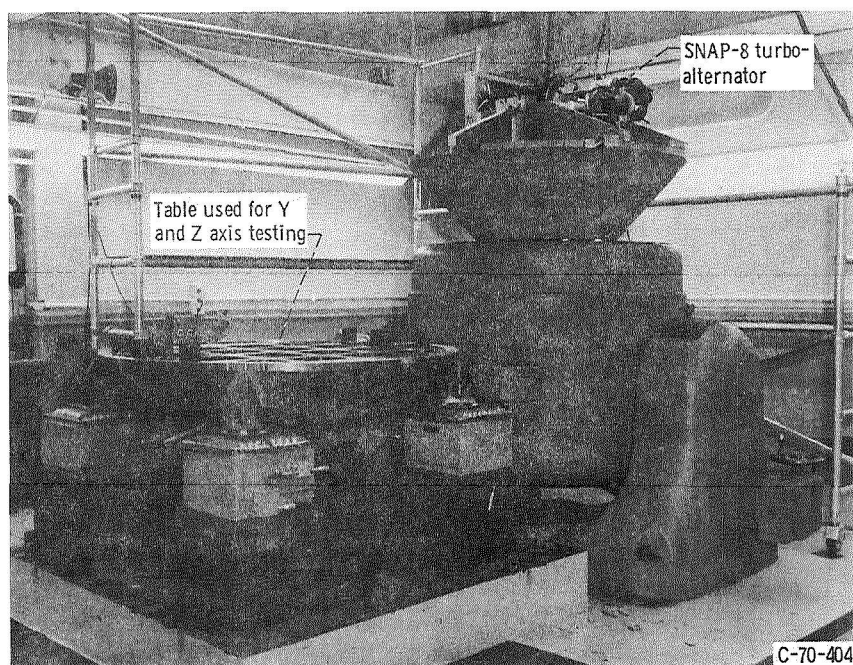


Figure 2. - MB electronics C210 vibration and shock machine in position for X axis testing.

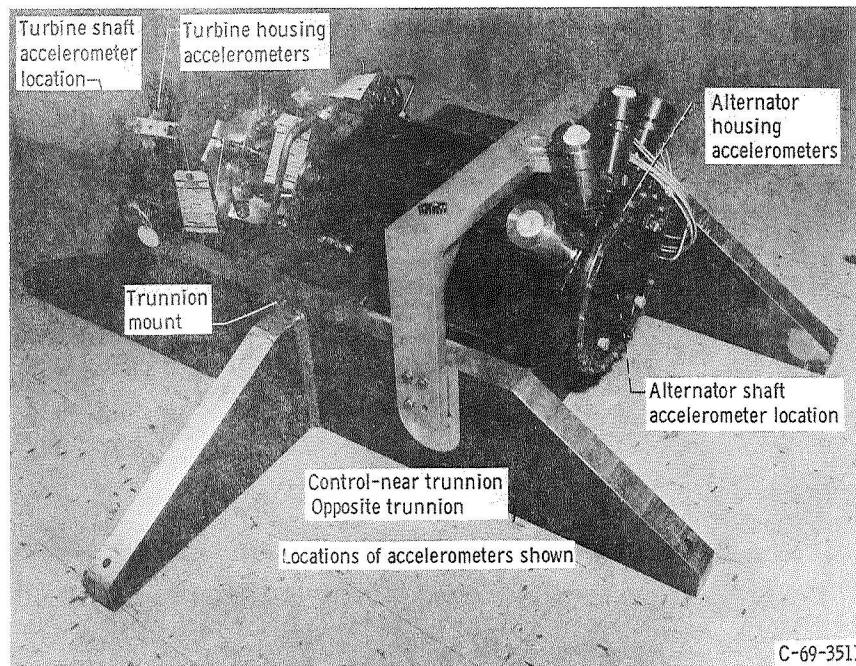
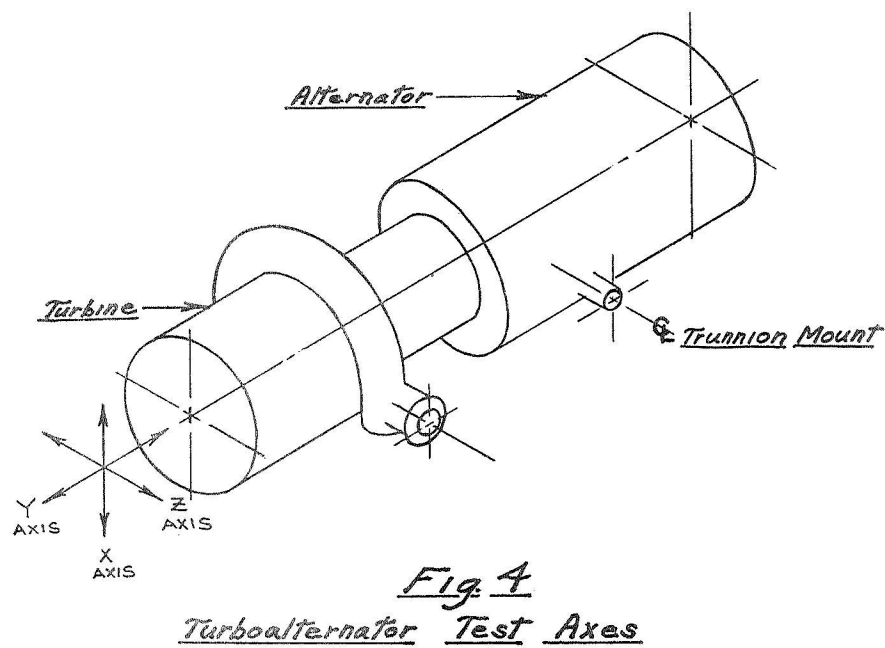


Figure 3. - SNAP-8 TAA turboalternator mounted in vibration test fixture.



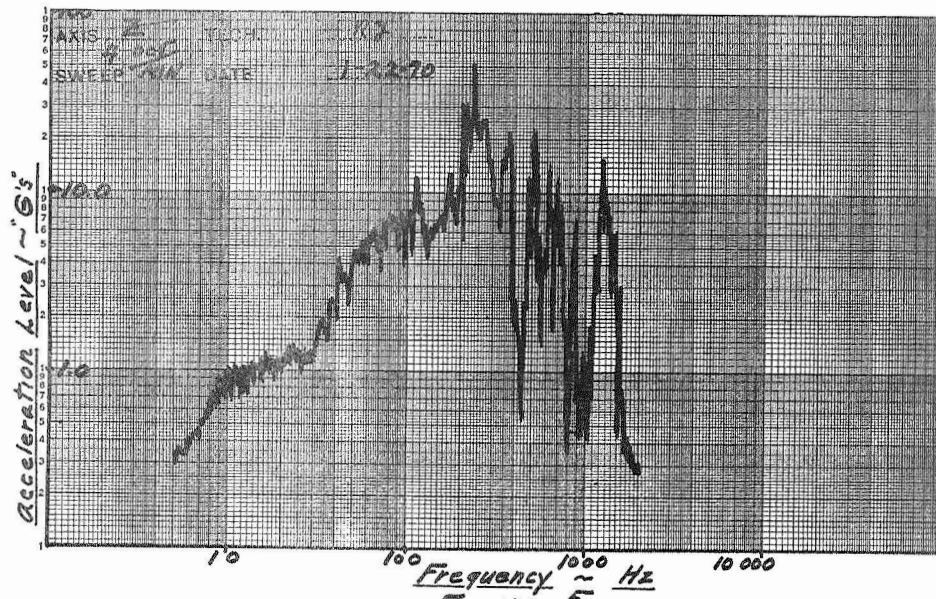


Figure 5
Response of Alternator Shaft Accelerometer, Z Axis, Z Axis Input

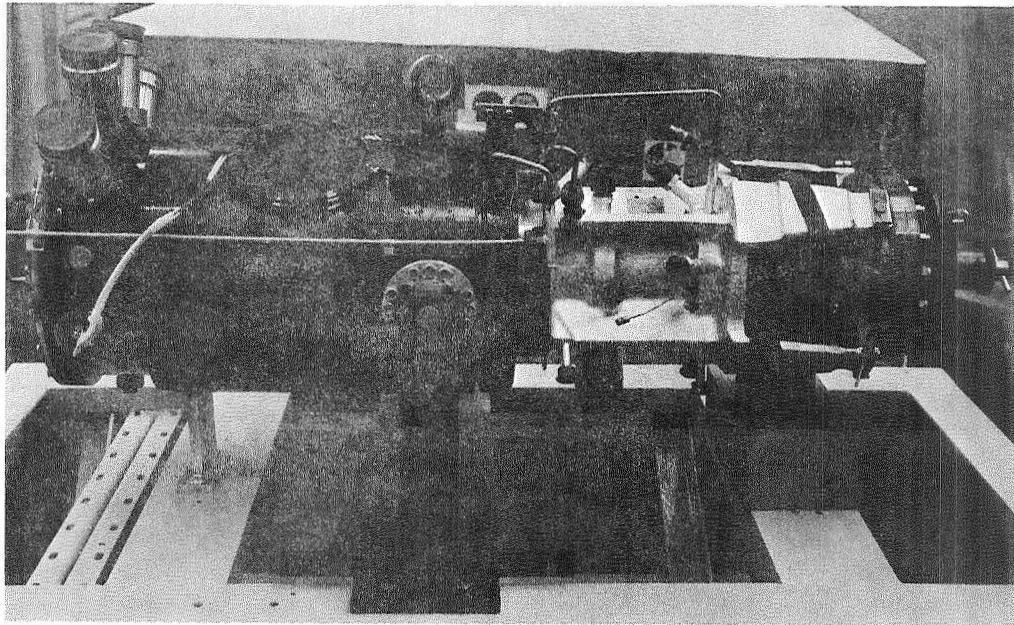


Figure 6. - SNAP-8 turbine alternator assembly unit 6/3 after vibration and shock testing, and upon receipt at A.G.C.

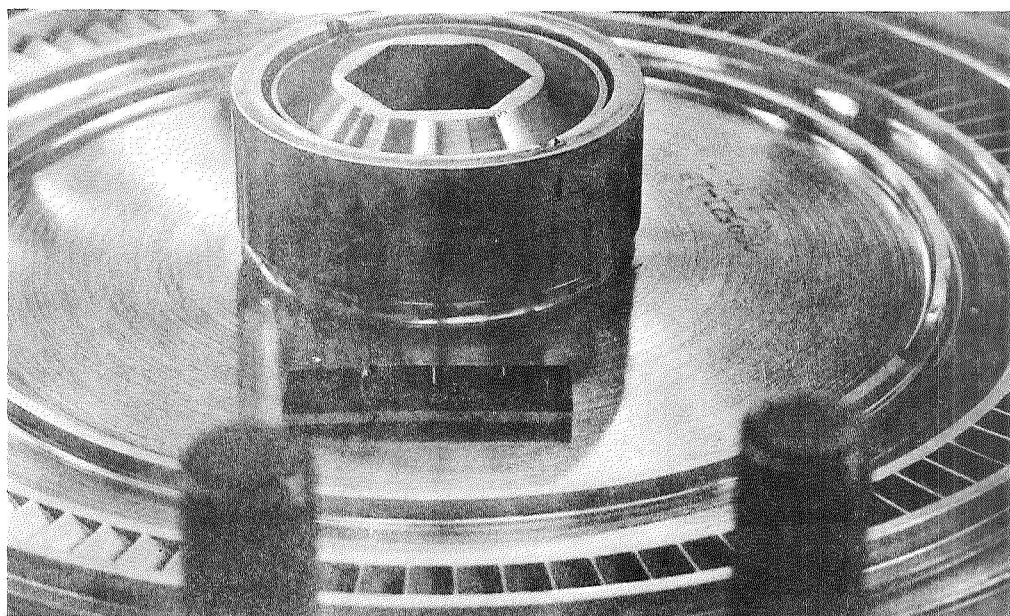


Figure 7. - First stage wheel close up of light labyrinth marking of outboard hub.

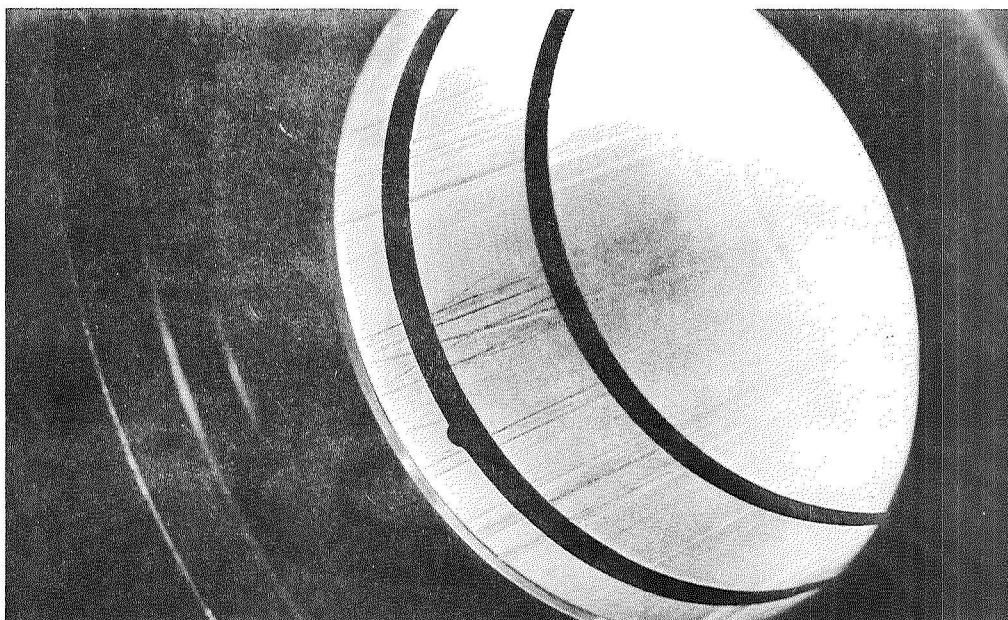


Figure 8. - Turbine bearing housing internal view showing gall marks from rotor removal.

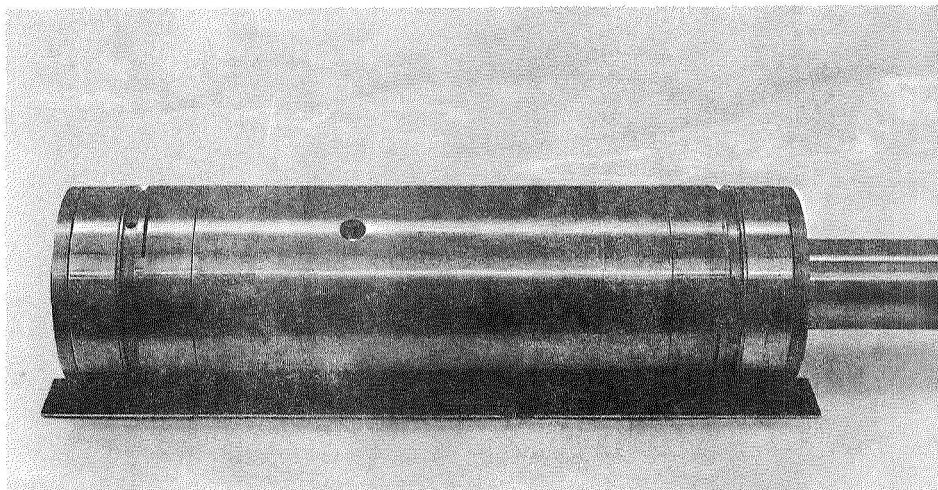


Figure 9. - Turbine shaft and bearing assembly.

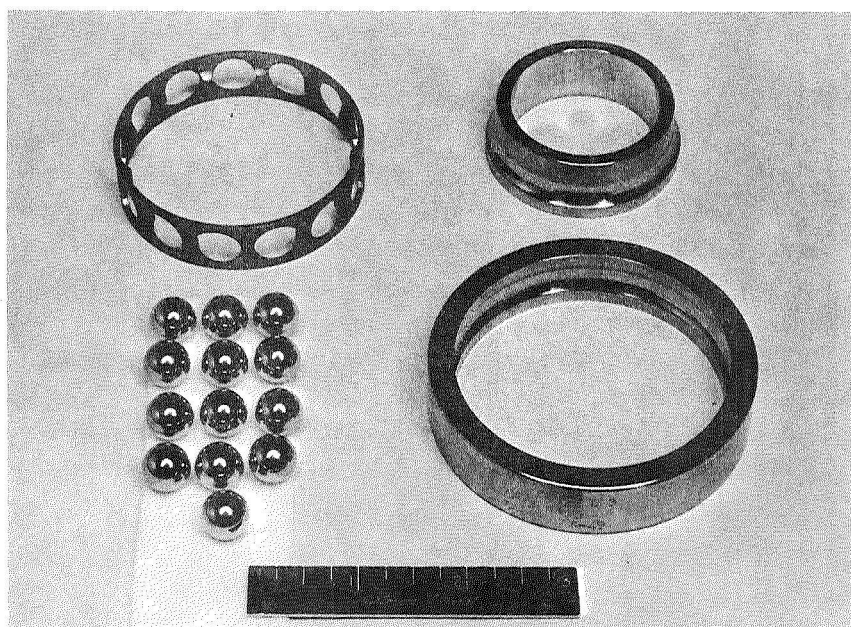


Figure 10. - Disassembled view of turbine bearing for the alternator end S/N A-49.

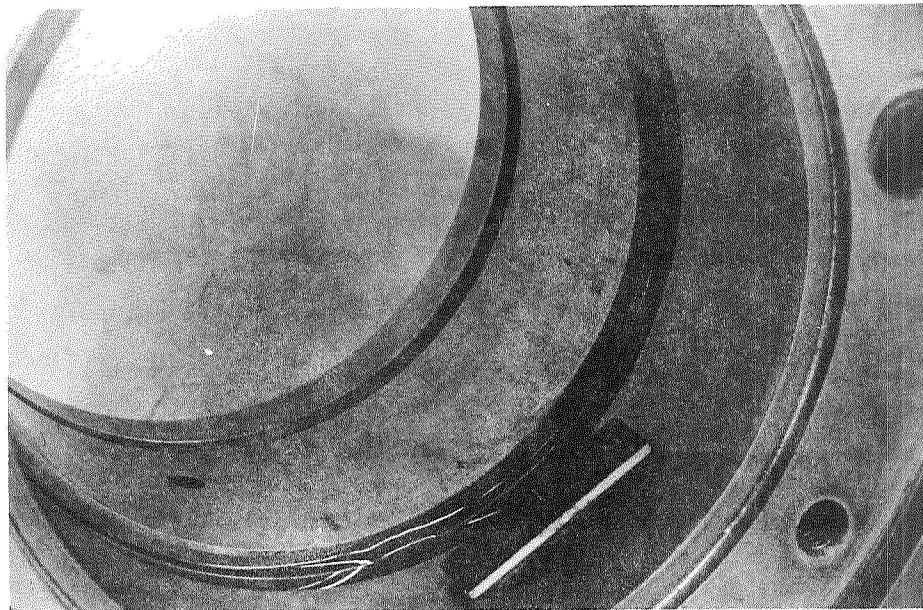


Figure 11. - Alternator antidrive end bell showing galls and scratches in bearing seat.

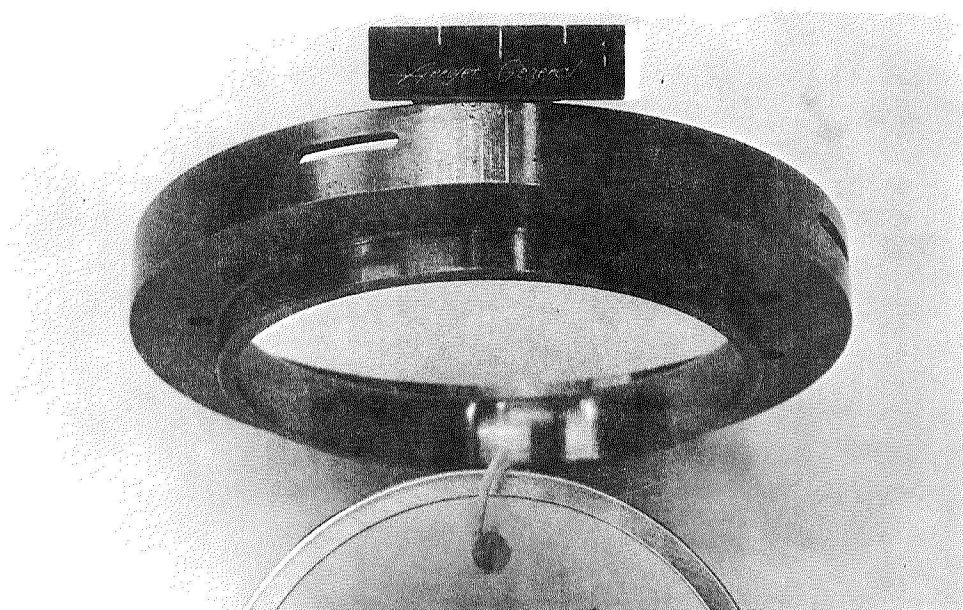


Figure 12. - Alternator antidrive end bearing stop showing galls and scratches.

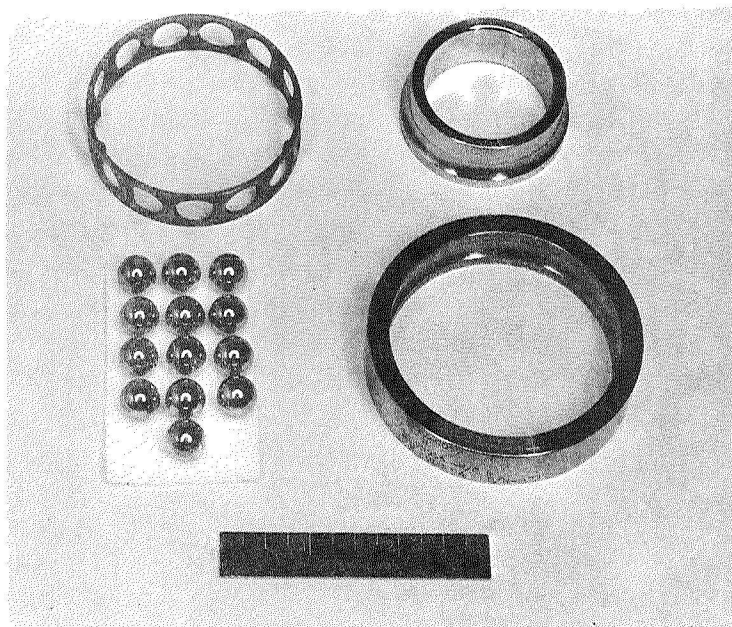


Figure 13. - Disassembled alternator drive end bearing S/N A73.

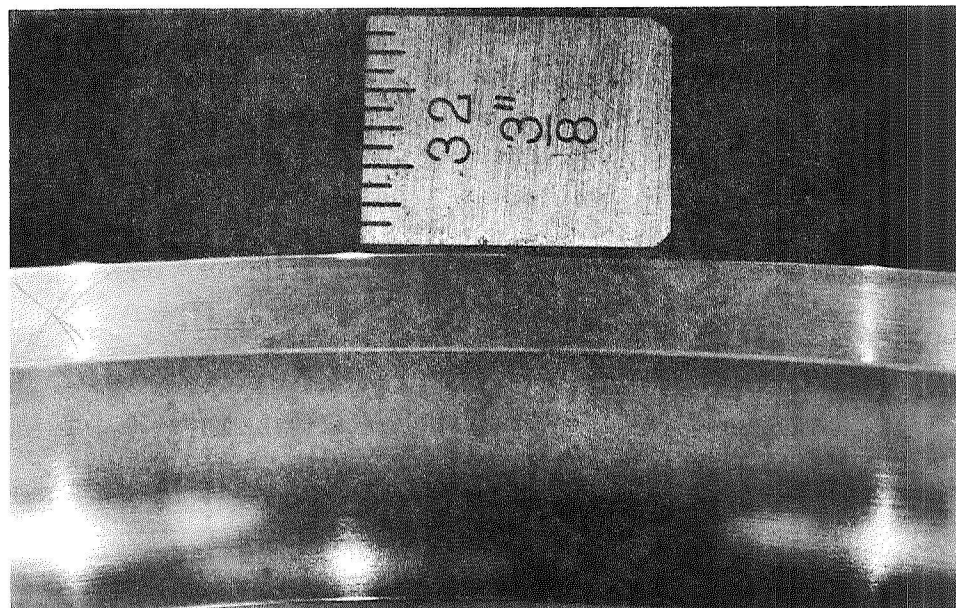


Figure 14. - Alternator antidrive end bearing outer ring raceway-S/N A74.